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ME 4182
Group 2

**Boom and Chassis
Articulation Joints**

June 4, 1992

Presented to:
Mr. James Brazell

by:
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(NASA-CR-192059) BOOM AND CHASSIS
ARTICULATION JOINTS Final Report
(Georgia Inst. of Tech.) 30 p

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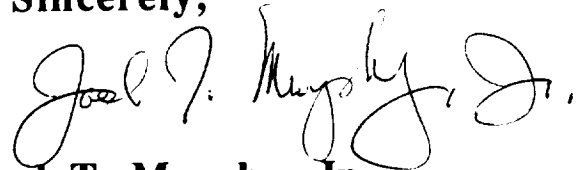
June 4, 1992

**Mr. James Brazell, Instructor
G. W. Woodruff School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, Georgia 30332**

Dear Instructor Brazell:

We are transferring the final report of our Me 4182 project. We would like to thank you for your support and assistance which has enabled us to reach our goal.

Sincerely,



**Joel T. Murphy, Jr.
Group Leader**

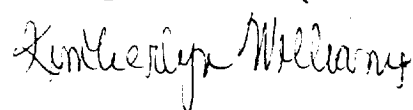


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2.1.1 ABSTRACT

The primary goal of our design project was to develop articulation joints for the chassis and boom of the proof-of-concept lunar vehicle. This is an ongoing project and the work of previous student groups was extensively reviewed. Some of the ideas generated are variations of past proposals. Although the project is funded by NASA/USRA, it is totally a student design effort.

2.1.2 Problem Statement

The design of the boom and chassis articulation joints are to be further developed for a proof-of-concept model of a lunar vehicle. The vehicle should be able to lift and move a massive object on the surface of the moon. Proper design will minimize the weight of the motors in the articulation joints and determine the optimum motor location. The design of the boom should have the lowest possible center of gravity.

Performance Considerations

- 1. Boom must support 10 pounds at maximum extension without tipping.**
- 2. Drives must allow full rotation of joint.**
- 3. Drives must be self-locking.**
- 4. Drive weight must be minimum.**
- 5. Drives must allow cable space through joint.**

2.2 Drive Selection

The process of choosing the joint drives was the most involved part of the project. The exact requirements were not known at the beginning of the quarter. As other groups designed their parts, information that had been previously estimated was provided and the calculations could be finalized. The dimensions of the vehicle were some of the requirements known at the start of the project. The outer diameter of the boom was stated as being 12" and the outer diameter of the chassis sections were considered to be 18". The other known requirement was that the boom had to be able to hoist a 10 pound weight. A decision was made by all the groups to use hydraulic power to operate the designs this quarter. All motors and equipment mentioned will be hydraulic operating at an average pressure of 2400 psi.

2.2.1 CHASSIS

For the chassis, weight was not a major concern as in the boom sections. A simple design was considered. It consisted of a motor that drives which, in turn, drives the joint. The gear is mounted to a Kaydon bearing on one side of the joint and is mounted to the frame of the chassis on the other side. The motor has 0.5 Hp to handle the maximum torque requirement of 29,000 in-lbs. There are six joints in the chassis. Four of these joints are perpendicular to the chassis axis while the other two are rotated 14.5° off the axis. This angle was optimized by the 4192 joint group. The angle requires some modification to the initial design. First the drive shaft must have a universal joint to connect the motor to the drive gear. Also, the gear mounted to the driven section will be placed in the center of the elliptical cross-section created by the 14.5° angle. For both joint types a speed reduction of 63:1 had to be used.

2.2.2 ARTICULATION JOINTS

The boom joints' operation is highly dependent on the weight it has to move. So, minimizing the weight of

5

the joints and the drives was critical. Many ideas were considered. The final design is based on a worm gear rotator produced by Gear products, Inc. It is driven by a 0.5 Hp Model 22B motor from Webster hydraulics. The design is simple and requires no additional moving parts. It has the capability of producing a torque of 32,000 in-lb and weighs only 48 lbs. which is comparable to other design options. Another merit for this design is its compactness. The entire assembly will fit into a cross-section of the boom with a maximum width of 2".

2.3 MOTOR ANALYSIS

2.3.1 ARTICULATION JOINTS

Hydraulic power is used to drive all joints. Important criteria for motor selection included existing technology to construct the design, lightweight, self-locking drives and ability to produce 32,000 in-lbs of torque. The alternatives proposed along with the disadvantages and advantages are as follows:

ALTERNATIVE I: (Figure 1)

HYDRAULIC CYLINDER DRIVING A GEAR THAT DRIVES THE JOINT

ADVANTAGES

- *Technology exists to construct design**
- *Lightweight**
- *Able to produce the torque needed**

DISADVANTAGES

- *Drives are not self-locking**

ALTERNATIVE II: (Figure 2)

HYDRAULIC CYLINDER DRIVING THE JOINT

ADVANTAGES

- *Technology exists to construct design**
- *Able to produce the torque needed**

DISADVANTAGES

- *Too heavy**

ALTERNATIVE III: (Figure 3)

ROUND HYDRAULIC CYLINDER DRIVING THE JOINT

ADVANTAGES

- *Lightweight**
- *Able to produce the torque needed**

DISADVANTAGES

- *Attempts to design a leak-free seal have been unsuccessful**

2.3.2 CHASSIS

Although the same hydraulic motors used in the articulation joints are used in the chassis, the criteria for selection differs from the criteria that applies to the articulation joints. Where as lightweight motors are a major concern for the articulation joints, the opposite is true for the chassis. The motors must provide enough weight to prevent tipping.

Placing a gear mounting in locations where the joints are not aligned at all times poses a problem. The problem involves the lifting of the pinion off of the race during rotation of the joint. To alleviate this problem, we have decided to mount a circular track around the joint as opposed to using a gear mounting.

Gear Design

At first, the gear was designed to be the internal gear system which is mounted to the inside wall of the boom., but there is a problem with this design is that the gear is not self-locking. This will help to prevent the situation when the hydraulic valves does not work properly or anything goes wrong with motor, the self-locking system wil keep the boom in place.

In choosing the proper self-locking gear, the design uses a totally enclosed a pre-lubricated 170 Series rotation system. This system is powered by a 6-cir Model RS motor from White Hydraulics, the self-locking rotator in corporated a hardened-steel worm an worm-gear slew ring. there is no requirement for backlash adjustment since it has true worm-gear tooth form, and full-tooth contacting that minimizes backlash. The gear has a ratio speed of 63:1 with a torque of 32000 in.lb

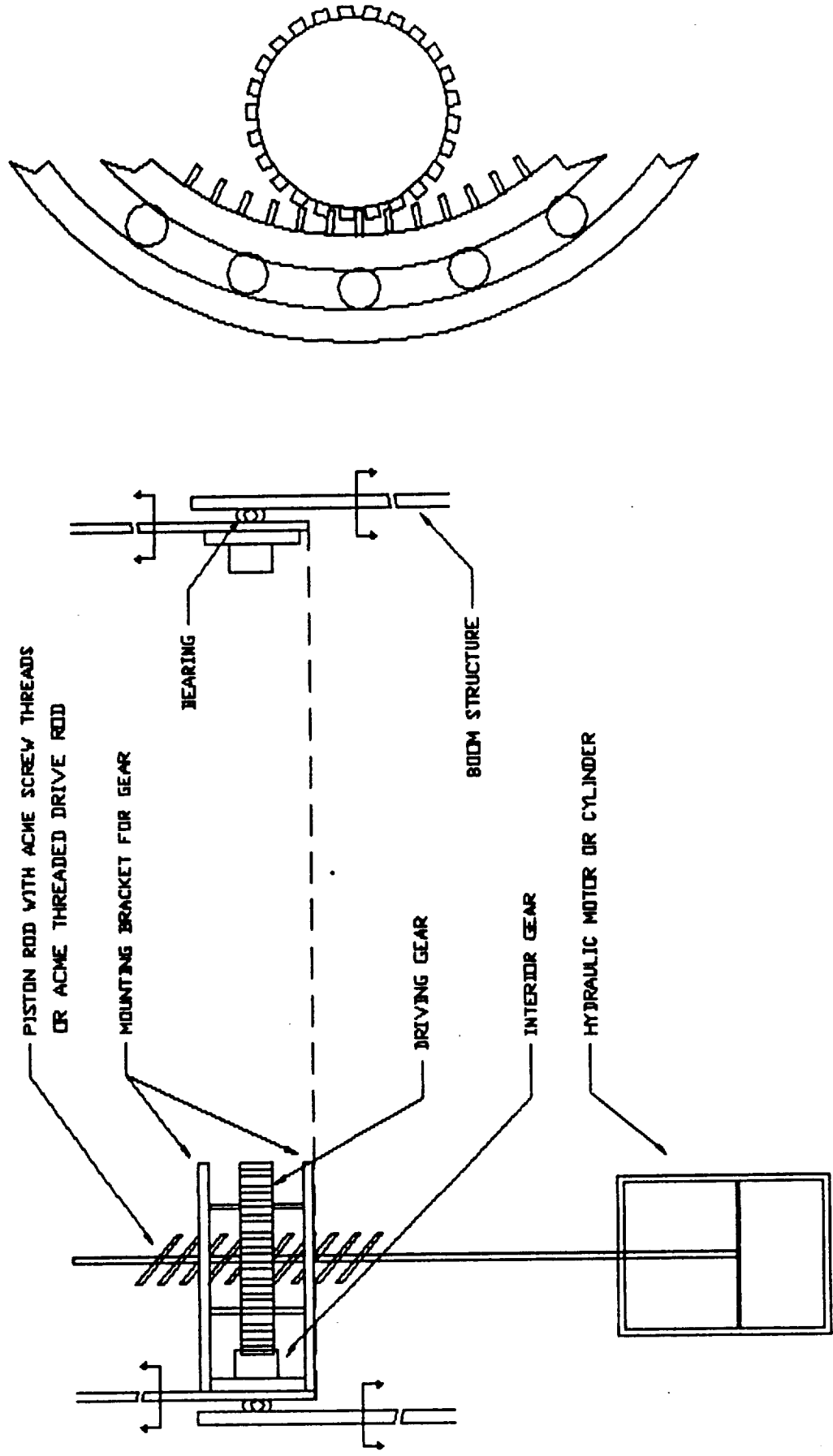
2.5 OVERALL DESIGN

The following list provides an itemized description of the final motor design for the articulation joints and chassis. One motor is housed in each articulation joint, with the exception of joint 2 which houses two motors primarily to prevent offsetting the center of gravity of the boom.

MOTOR TYPE:	Model M22B
SPEED:	500 RPM
INPUT FLOW:	0.6 GPM
PRESSURE:	2400 PSI
HP OUTPUT:	0.508 HP
WEIGHT:	20 LBS
MANUFACTURER:	MacMillin Hydraulic Engineering Corp. & Rexrothe Corp.

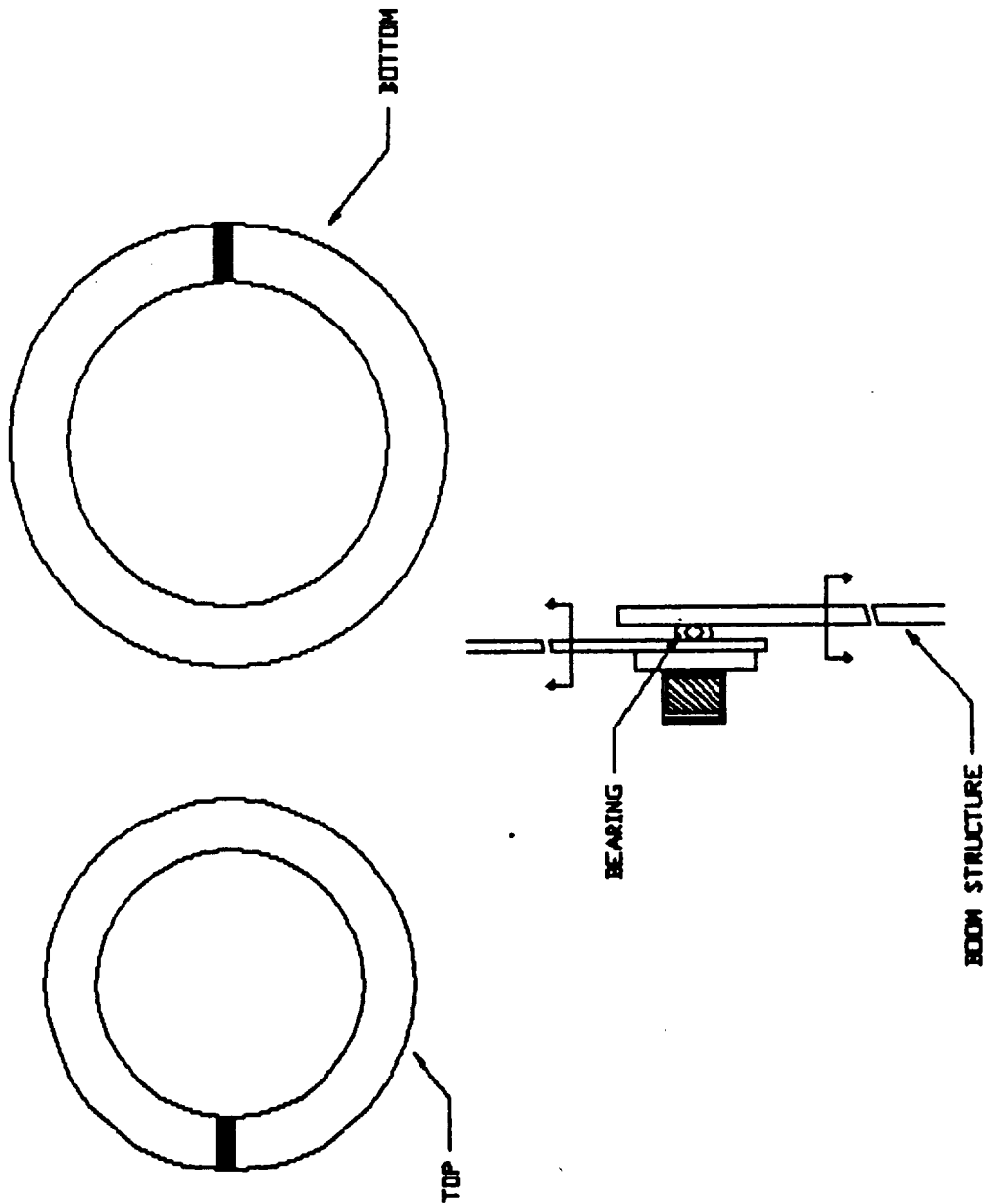
APPENDIX I

HYDRAULIC CYLINDER DRIVING a GEAR THAT DRIVES THE JOINT

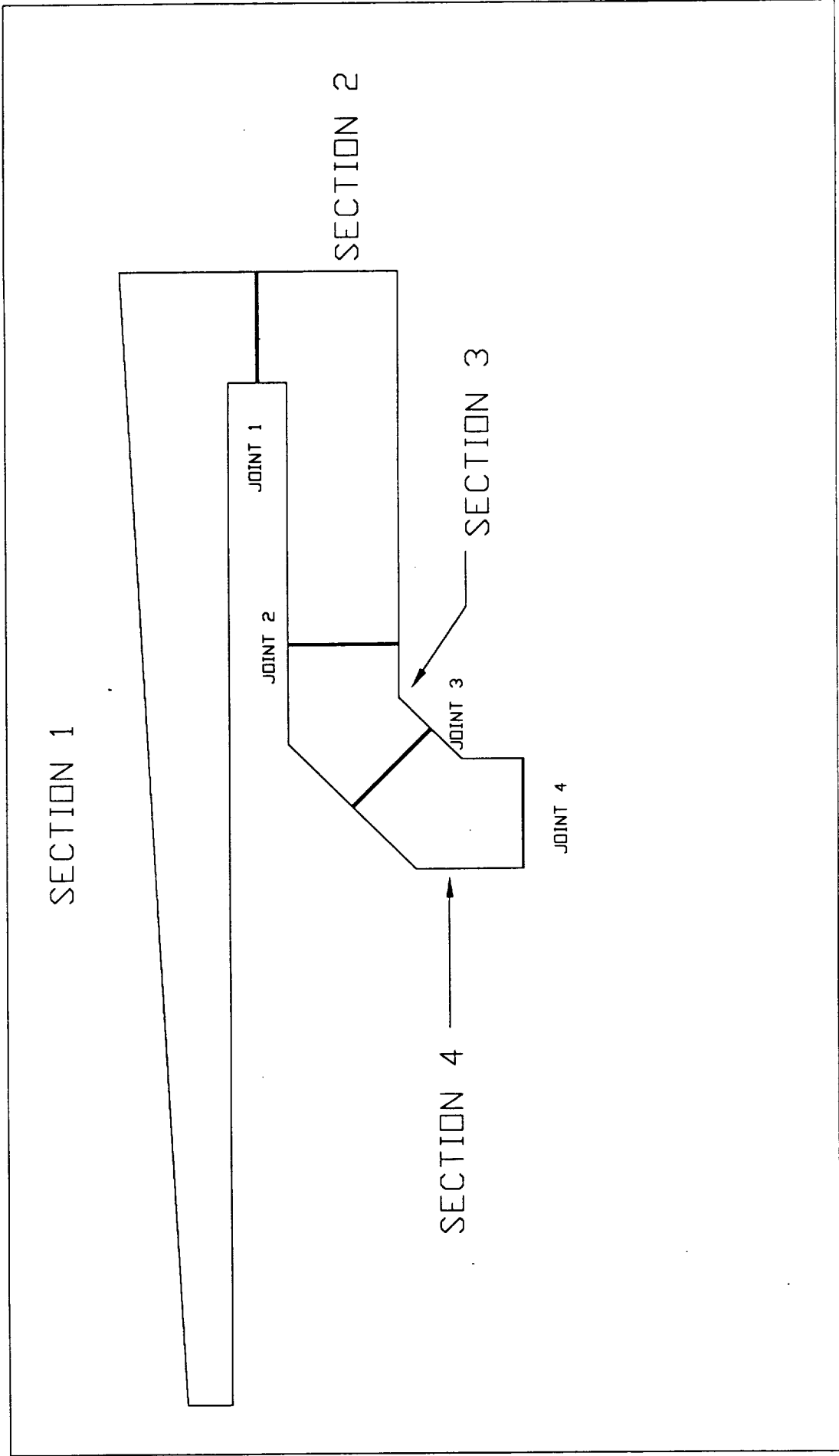


ALTERNATIVE 1
(1.1.1)

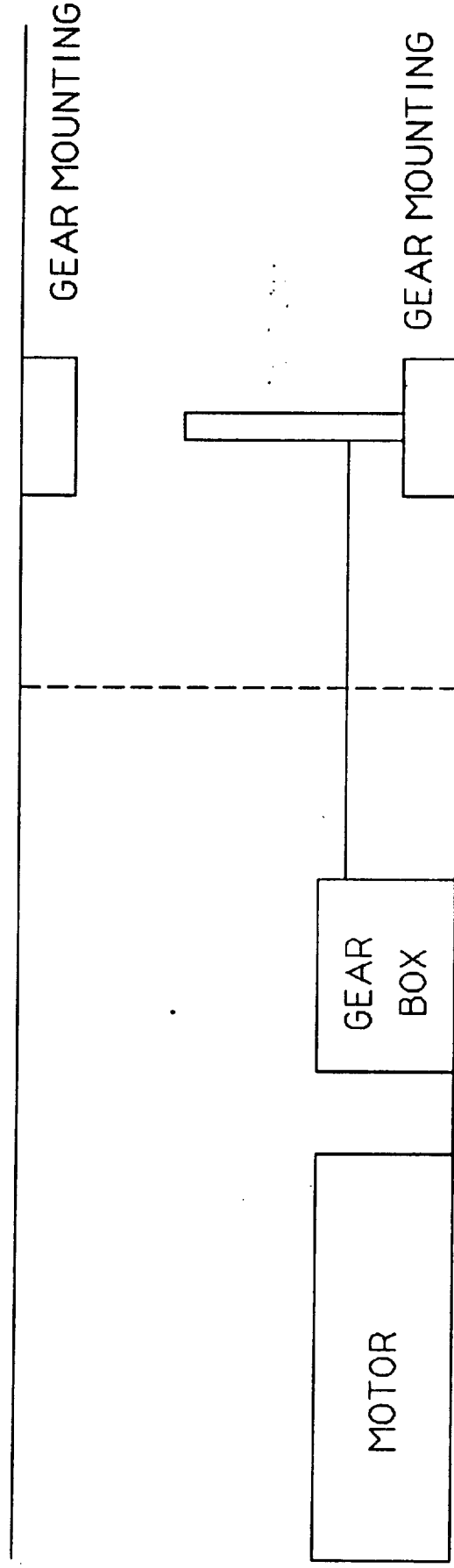
ROUND HYDRAULIC CYLINDER DRIVING JOINT



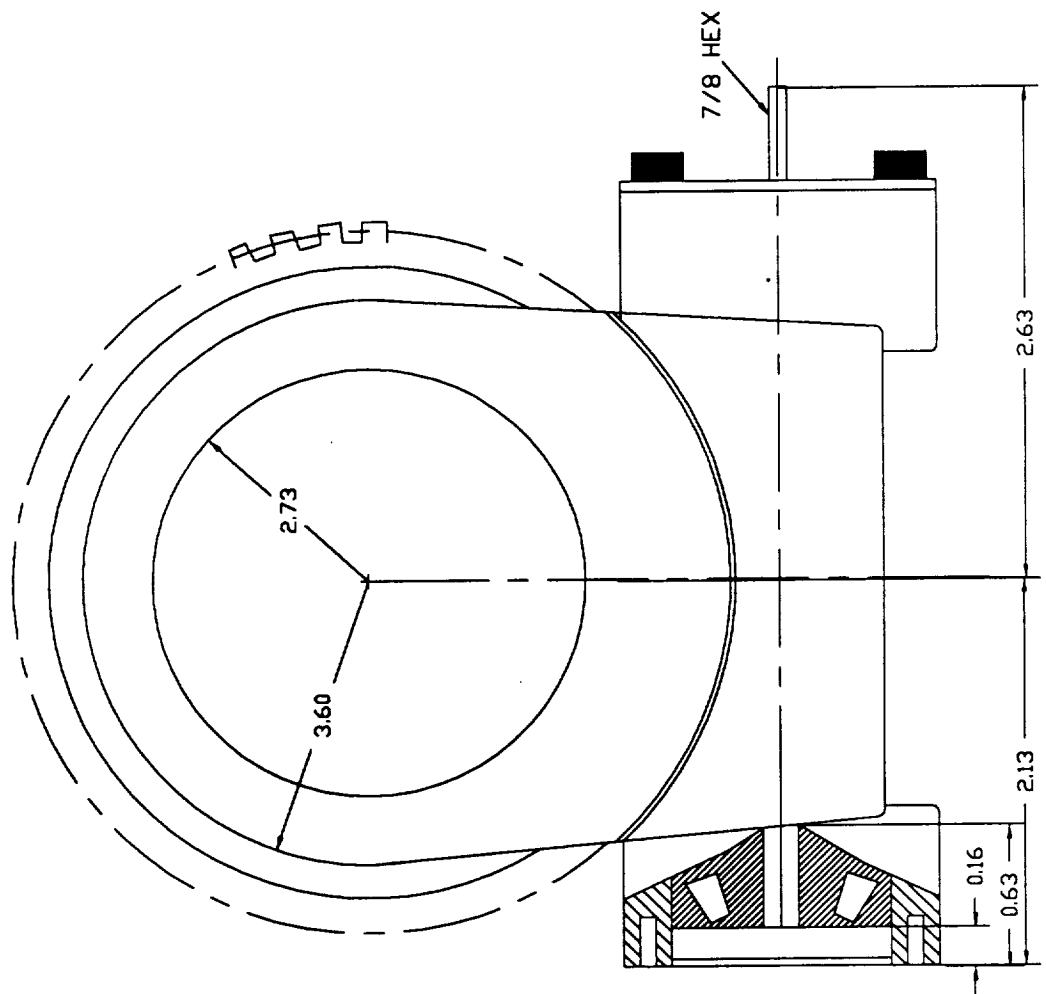
ALTERNATIVE 3
(Fig. 3)

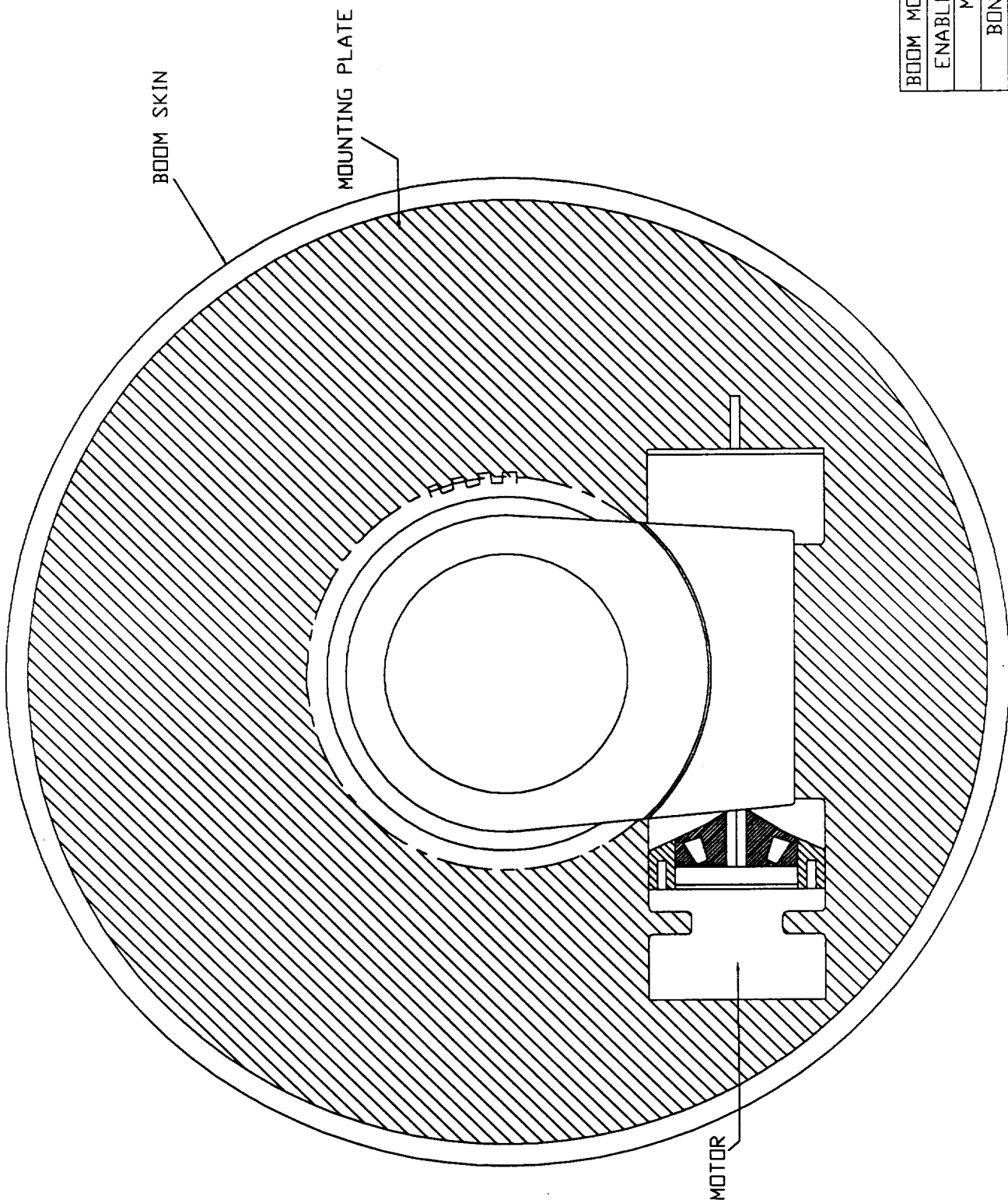


CHASSIS JOINT

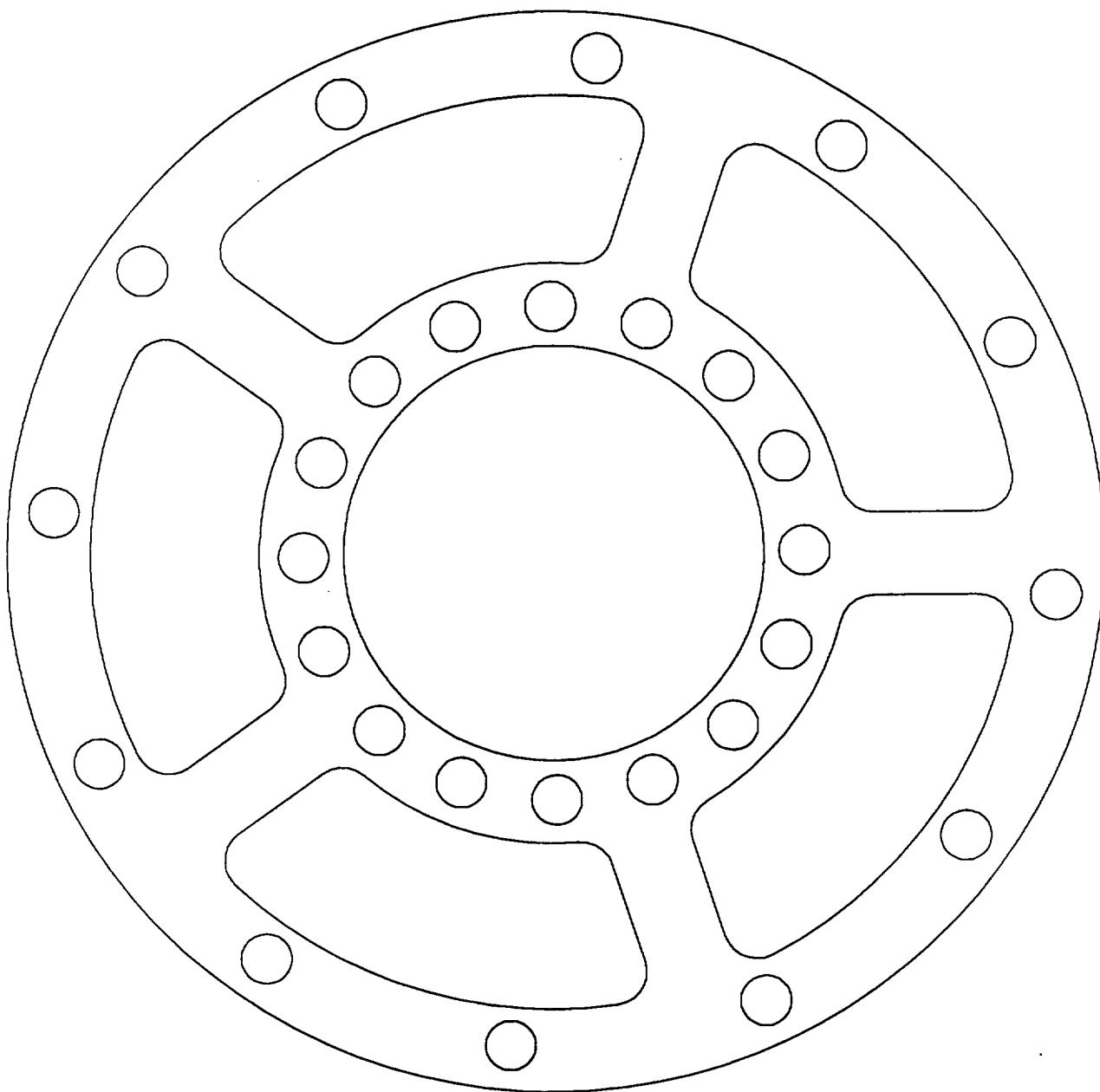


BOOM JOINT DRIVE DETAIL
ENABLER
MAY 30, 1992
BONNIE L. TURNER

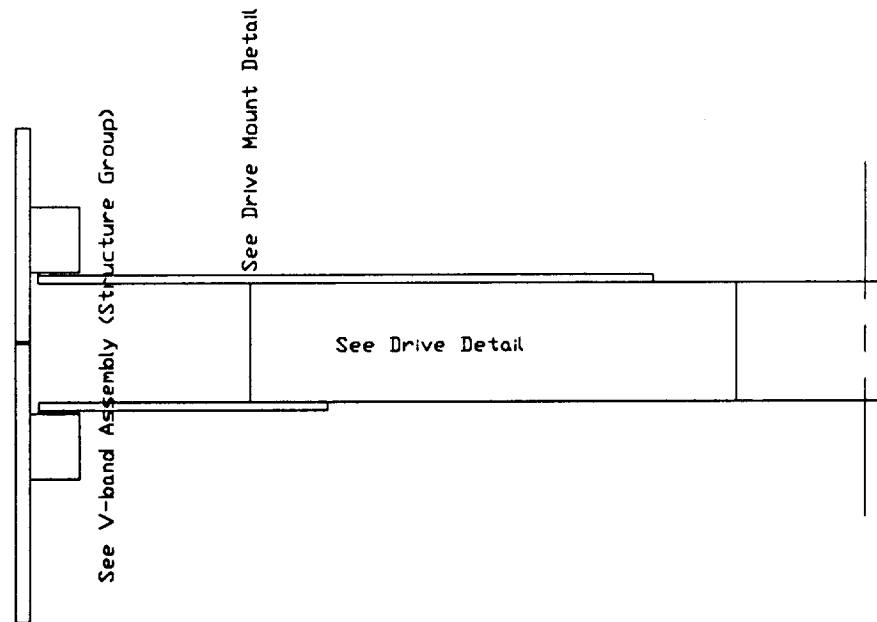




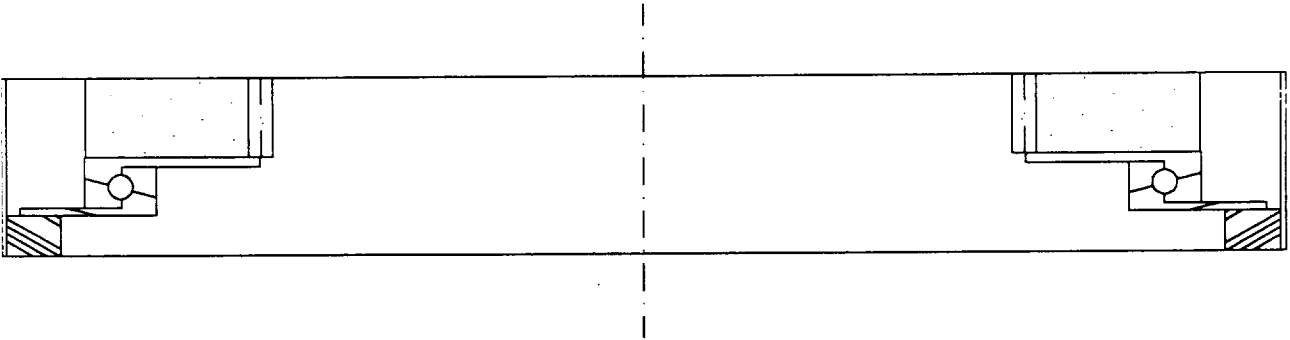
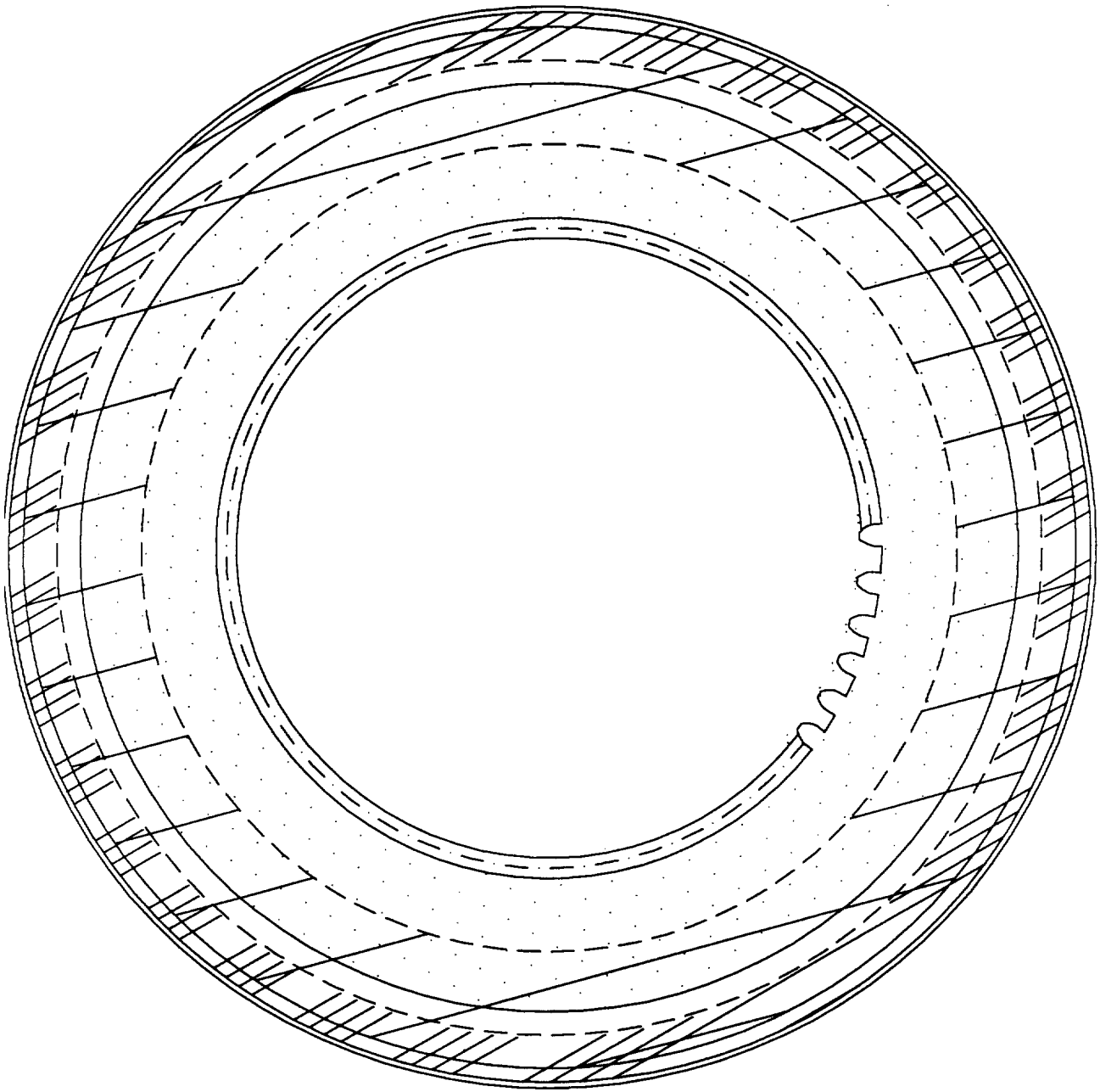
BOOM MOTOR MOUNT ASSEMBLY
ENABLER —NOT TO SCALE
MAY 30, 1992
BONNIE L. TURNER

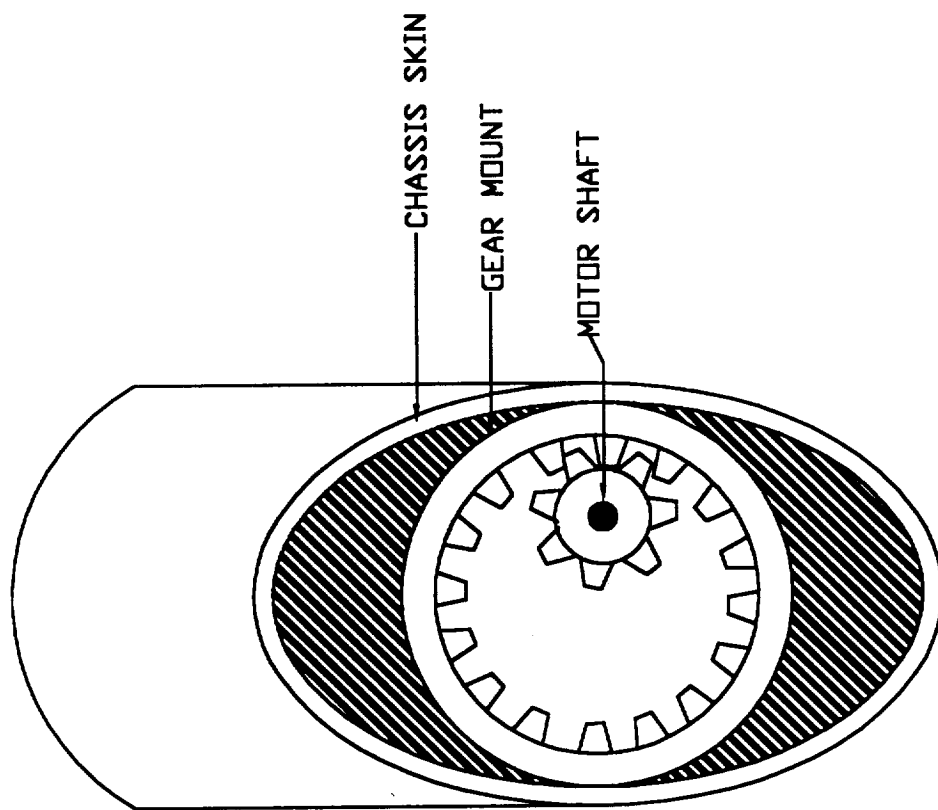


BOOM MOTOR MOUNTING PLATE
ENABLER
JUNE 2, 1992
BONNIE L. TURNER



Boom Joint Detail
Enabler
5-29-92
Joel T. Murphy Jr.





NOTE: NOT TO SCALE

CHASSIS MOTOR GEAR CONNECTION

ENABLER

JUNE 1, 1992

BONNIE L. TURNER

APPENDIX II

Calculations of Moment at each Joint

A. Moment at joint #1

$$M = 10 * 16.42 + 87.37 * 6.84 = 761.81 \text{ lb ft}$$

$$A = 10 + 87.37 = 97.37 \text{ lbs}$$

$$S = 0$$

B. Moment at joint #2

$$M = 10 * 24.3 + 87.37 * 8.89 + 51 * 2.05 = 1124.27 \text{ lb ft}$$

$$A = 0$$

$$S = 51 + 87.37 + 10 + 31.50 = 179.87 \text{ lb}$$

C Moment at joint #3

$$M = 10 * 24.3 + 87.37 * 14.72 + 51 * 9.98 + 31.5 * 5.83 = 2221.71 \text{ lb ft}$$

$$A = 246.34 \text{ lbf}$$

$$S = 246.34 \text{ lbf}$$

D Moment at joint #4

$$M = 8.44 * 316.71 = 2672.16 \text{ lb ft}$$

$$A = 316.71 \text{ lbf}$$

$$S = 0$$

Bolt Analysis

F_t = Total force exerted on all bolts

$$= M / (4 * R)$$

Where R = radius of shear plane.

$$F_t = 2672.16 / (4 * .5) = 1336.08 \text{ lbf}$$

F_b = Force exerted on each bolt

N = Number of bolts used

$$= 12$$

$$F_b = F_t / N = 11.34 \text{ lbf/bolt}$$

T = Shear stress on one bolt

A_b = Cross-sectional area (nominal) of one bolt

$$= .2245 \text{ in}^2$$

$$T = F_b / A_b = 496 \text{ psi / bolt}$$

S_y = minimum shear yield strength (which must be greater than $12 * T$)

$$= 36 \text{ kpsi}$$

This is safe design.

Factor of Safety

$$\sigma = M * (I/c)$$

$$I/c = \pi * (d_o^4 - d_i^4) / (32 * d_o)$$

$$= \pi * (.3048^4 - .3033^4) / (32 * .3048) = 5.52E-5 \text{ m}^3$$

$$M = 2672.16 \text{ lb ft} = 3622.97 \text{ N m}$$

$$\sigma = 6.57 \times 10^7 \text{ Pa}$$

$$S_y = 170 \text{ MPa}$$

$$\text{Factor of safety} = S_y / \sigma = 2.59$$

Power required

$$\text{Torque} = F * r$$

$$T (\text{Maximum}) = 32064 \text{ lb in}$$

$$\text{Horse Power} = (T * \text{rpm}) / 63000$$

$$\text{rpm} = \text{Speed of articulation joint} = 1 \text{ rpm}$$

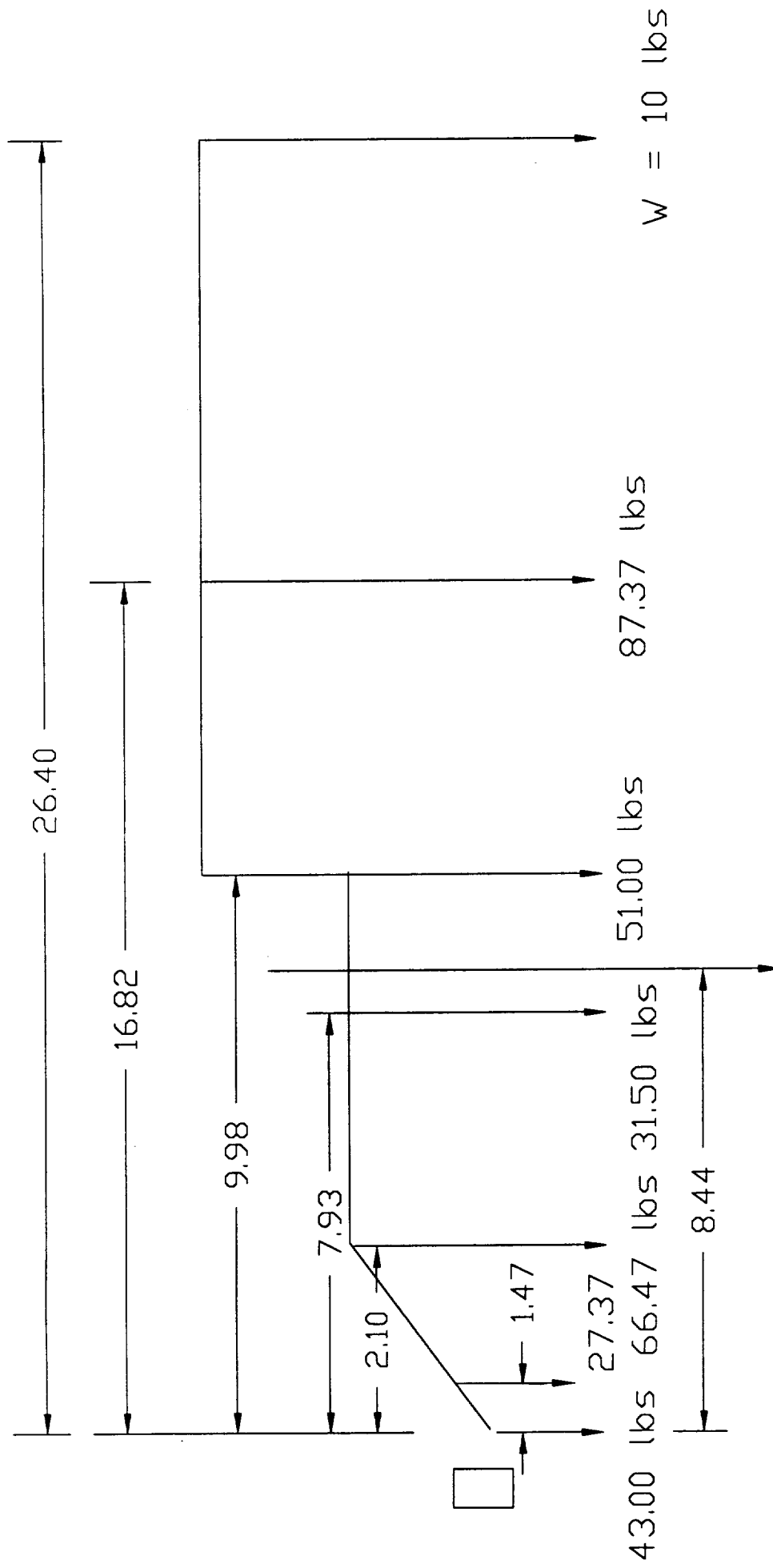
$$\text{Horse power required} = .508$$

BOOM VOLUME

DIM (m)					
	<i>LENGTH</i>	<i>DIAMETER 1</i>	<i>DIAMETER 2</i>	<i>ELBOW VOL</i>	<i>MAT. THICK</i>
SECT 1	5.0952	0.15	0.3	0.025446879	0.03
SECT 2	1.774	0.3	0.3	0.025446879	0.03
SECT 3	0.2149	0.3	0.3	0.20478086	0.03
SECT 4	0.5	0.3	0.3	0.20478086	0.03
C. OF G.					
SECT 1					
SECT 2					
SECT 3					
SECT 4					

BOOM VOLUME

<i>OUTER VOLUM</i>	<i>IN VOL</i>	<i>TOTAL VOL</i>	<i>BOOM VOL</i>	<i>ACT. WEIGHT</i>
0.21009226	0.142862737	0.092676402	0.591019739	
0.125396565	0.080253802	0.070589642		
0.015190373	0.009721839	0.210249394		
0.035342888	0.022619448	0.2175043		



CENTRAL GRAVITY OF THE BOOM

$$\begin{aligned}
 X * (\text{TOTAL WEIGHT}) &= \text{SUM OF } X_i * \text{INDIVIDUAL WEIGHT} \\
 X * 361.71 &= 10 * 26.40 + 87.37 * 16.82 + 51 * 9.98 + 7.93 * 31.5 \\
 &\quad + 2.10 * 66.47 + 27.37 * 1.47 \\
 &= 2672.16 \text{ LB-FT} \\
 X &= 8.44 \text{ FT FROM PIVOT}
 \end{aligned}$$

